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Title:

Evaporation process control used in refrigeration

Technical field:

Evaporation of refrigerant in cooling and freezing plants, refrigeration engineering, refrigeration machine for cooling and heating operation, refrigeration plants, refrigeration sets, heat pumps, air-conditioning systems and others.

Prior art:

Evaporator control with drive dry expansion on the basis of the minimum stable signal (MMS) (Fig. 1, 2 and 3).

For optimum operation of an evaporator used in refrigeration, the evaporator is supplied with sufficient wet steam for a control valve (expansion valve) (3) to be controlled to a minimum stable signal, normally on the basis of the evaporator outlet pressure (12) and the associated evaporator outlet temperature (13) (drawing Fig. 1, 2 and 3). The difference between the evaporator pressure, converted by calculation into the associated evaporation temperature, and the actual evaporation temperature measured is used as measured variable for the control valve. In this context, the aim is stable control characteristics with as low a temperature difference as possible. As low a temperature difference as possible leads to a higher evaporator power. If the difference is too low or the signal is not stable, liquid shocks or power reduction occur at the compressor (1). If the difference is too great, the evaporator power (4) is reduced.

Automatic valves, capillary tubes or other equipment are also dimensioned and used on the basis of the same

principle (superheated refrigerant vapor at the end of the evaporation process).

Nowadays, in some cases internal heat exchangers (IHEs) (5) (Fig. 4, 5, 6) are connected downstream of the evaporator. However, these internal heat exchangers are designed as "thermally short" equipment and are not incorporated in the evaporator control on the basis of the entry vapor content. The refrigerant liquid is not strongly cooled and the suction vapors are not strongly superheated. The superheating of the suction vapor is restricted to approx. 5-10K. Injection valves which are customary nowadays are also not designed for maximum superheating, and the superheating which can be set is at most approx. 20-25K.

Detailed description of the invention:

It is an object of the invention to achieve the following in cooling/freezing plants, refrigeration machines for cooling and heating operation, refrigeration plants, refrigeration sets, heat pumps, air-conditioning systems and all other systems using refrigerant for evaporation:

To keep the suction vapor superheating in the evaporator (4) at a low level or to leave the evaporator (4) with wet steam, and in this case keeping the suction vapor superheating upstream of the compressor (1) as high as possible (as far as the use limits of the compressor, the oil or the refrigerant and/or the various temperature ratios permit).

For this purpose, the refrigeration plant, which primarily comprises compressor (1), condenser (2), injection valve (3) and evaporator (4), is provided with an additional internal heat exchanger (5), referred to below as IHE (Fig. 7, 8, 9, 10, 11).

This IHE (5) is installed between evaporator (4) and compressor (1), on one side, and between condenser (2) and injection valve (3) on the other side (drawing Fig. 8, 9, 10).

On one side, liquid refrigerant flows through the IHE (5) (liquid side), and on the other side superheated refrigerant in vapor form or wet steam flows through the IHE (5).

If pure media (liquid refrigerant and superheated suction vapor) flow through the IHE, it is possible to speak of heat exchange (Fig. 4, 5, 6). If the IHE is operated with a liquid refrigerant and wet steam with subsequent suction vapor superheating, it is possible to speak of a second evaporation stage with integrated liquid supercooling and suction vapor superheating (Fig. 7, 8, 9, 10). The following text always encompasses both possible options.

The actual evaporation (first stage) (4) takes place partly or completely in the evaporator (4). To allow optimum operation of this evaporator (4), liquid refrigerant is admitted at the evaporator outlet.

Since liquid refrigerant is admitted at the evaporator outlet, for control of the evaporator (4) there is an absence of a measurement variable for determining the superheating, and the expansion valve (3) can no longer control the filling of the evaporator (4) with refrigerant.

The control for which a patent is hereby applied for the first time, as a novel feature, makes use of the measurement variables comprising the liquid temperature of the refrigerant upstream of the injection valve (3) and the evaporator pressure (Fig. 7, 8, 9, 10, 11, points 9, 10, 11, 12).

It is in this context irrelevant what types or designs of evaporators and what refrigerants and application areas are involved.

The evaporator pressure is preferably taken at the inlet of the evaporator (12) (start of evaporation) (Fig. 7, 8, 9, 10, 11, point 12). In special cases, the exit pressure or any desired value derived from the two pressure measured values (refrigerant glide) can also be used as measured value (Fig. 7, 23).

This control controls the start of the evaporation process (Fig. 7, points 11, 12) rather than, as has hitherto been the case, the end of evaporation (Fig. 3, points 12 and 13).

It is in this context irrelevant whether control is set to precisely the left-hand limit curve between refrigerant liquid and refrigerant wet steam in the lg p, h diagram of the refrigerant or to a value (to the left) or to the right of this limit curve.

With "optimized" evaporator designs, the evaporation process is started as close as possible to the left-hand limit curve of the lg p, h diagram. In the case of non-optimized evaporators, it may be advantageous for a certain proportion of gas to be admitted at the start of the evaporation process. In this case, the evaporation process is started to the right of this limit curve after the optimum for the respective evaporator.

The start of the evaporation process can be defined by the liquid temperature upstream of the injection valve (11, 9) and the evaporation pressure (12, 10) (Fig. 7, 8, 9, 10, 11, points 11, 12, 9, 10).

The control variable can be defined, and the superheating controlled, from the evaporation pressure and a fixed (temperature) difference (adjustable) or from a stored curve calculation, depending on the refrigerant.
The injection valve (3) lowers the temperature of the refrigerant liquid (11) upstream of the injection valve (3) by opening the valve (3), and increases the refrigerant liquid temperature by closing the valve (3), thereby seeking to keep the desired value at a corresponding evaporation pressure (12).

The degree of flooding or superheating (19, 13) of the evaporator(s) (4) therefore determine the supercooling temperature of the liquid refrigerant (11) at a corresponding evaporation pressure (12) and the suction vapor temperature (13) at the compressor inlet (14).

When limit values are reached, such as for example the maximum permissible temperature for the compressor (13, 14, 15, 16), a further temperature-measuring sensor (optional) takes over and overcontrols the control of the refrigerant liquid entry temperature into the injection valve (11) on the basis of evaporator pressure (12) (Fig. 7, 9, 11, points 11, 12 and 13 (14, 15, 16)).

It is in this context irrelevant whether the suction vapor temperature at the exit of the IHE (5) (13), the suction vapor temperature at the compressor inlet (1) (14), the hot-gas temperature (compressor exit) (15), the oil temperature of the compressor (1) (16) or another suitable temperature is used as measurement variable for this safety and optimization function (Fig. 8, 9, 10, 11, points 13, 14, 15, 16).

In any event, an optimum-maximum supercooling (11) of the refrigerant liquid and an optimum-maximum suction

vapor superheating (14) as a function of the corresponding compressor is always the aim, as a function of the evaporator type (Fig. 7, 9, 10, 11, points 11, 14).

It is in this context irrelevant whether the refrigeration system comprises one or a plurality of evaporators (4), one or a plurality of IHEs (5), one or a plurality of compressors (1) or one or a plurality of injection valves (3), and whether or not they are combined to form groups. It is in this context also irrelevant whether or not one or more evaporators (4) are combined into groups with only one or more IHEs (5) (Fig. 10-18, points 9, 10, 13, 14, 15, 16). Any combinations of injection valves (3), evaporators (4), IHEs (5) and compressors (1) is therefore possible.

It is irrelevant whether the injection valves (3) are of mechanical, thermal, electronic or other design and whether they control cyclically, continuously or in some other way. What is crucial is the process and control circuit, with the dependent relationships which have been listed between start of evaporation (11, 12), end of evaporation (13, 19) as a function of the refrigerant liquid entry temperature (21) to the IHE (5), the suction vapor exit temperature (13) from the IHE (5), the state of the refrigerant (wet steam (19) or superheated suction vapor (13)) on leaving the evaporator (19) and entering (20) the IHE (5), which in one case is operated as a second evaporator stage with subsequent high suction vapor superheating (13) and in another case, in the same plant, is operated as a pure heat exchanger for superheating the suction vapor (13). In this context, it also irrelevant whether an external supercooler stage (25) connected upstream of the IHE (5) is connected to or disconnected from the process.

The advantage of this evaporator control consists in

the fact that in this way the evaporator (4) is optimally flooded and utilized (drawing Fig. 7, 9, 10, 11, points 17, 19), that the pressure drop on the refrigerant side across the evaporator (4) is reduced, that as a result the evaporation temperature (23) is increased, that as a result smaller evaporators (4) can be used, that as a result the mass flow of refrigerant for a required refrigeration power is reduced, that as a result the compressors (1) are smaller (refrigeration production), that as a result less energy is required for the generation of refrigeration, that as a result efficiencies and the lubrication and therefore the service life of the compressors (1) are increased.

The control is set in such a way that the maximum power is always in favor of the evaporator (4) (Fig. 7, 8, 9 points 17) and not the IHE (5) (18) (maximum possible enthalpy distance at point 17).

Novelty:

A novel feature of our invention is that an evaporation system with dry expansion is used as flooded evaporator (4), in which the refrigerant leaves the evaporator (4) in the first stage with liquid fractions (17, 19).

A novel feature of our invention is that the refrigerant enters a second evaporation stage (5, 18, 20) (dry evaporator) as a liquid/gas mixture with a high gas content, and residual evaporation with subsequent high superheating of the refrigerant (13) and simultaneous supercooling of the liquid refrigerant on the second side of the IHE (5) takes place in this second evaporation stage (11).

A novel feature of our invention is that control is based on the start of evaporation (12) of the evaporation process and not on the end of evaporation (13).

A novel feature of our invention is that this control is run on the evaporator (1) with different suction vapor superheating levels (13) depending on the liquid entry temperature (21) to the IHE (5).

A novel feature of our invention is that the suction vapor superheating (13) is selected to be as high as possible.

A novel feature of our invention is that the expansion valve (3) used, which is installed outside or inside the evaporator, controls the refrigerant liquid temperature (11) before it enters the injection valve (3).

A novel feature of our invention is that the expansion valve (3) used, which is installed outside or inside the evaporator (4), limits the suction vapor temperature at the entry to the refrigerant compressor (14) and at the same time controls the power of the internal supercooling (18) as a function of the evaporator power (17) available from the first stage (4).

List of drawings:

- Fig. 1 shows the refrigerant circuit in the "prior art" lg p, h diagram
- Fig. 2 shows the "prior art" refrigerant circuit
- Fig. 3 shows the refrigerant circuit in the lg p, h diagram with integrated equipment
- Fig. 4 shows the refrigerant circuit in the lg p, h diagram with IHE of the "prior art"

- Fig. 5 shows the refrigerant circuit with IHE of the "prior art"

- Fig. 6 shows the refrigerant circuit with IHE of the "prior art" in the lg p, h diagram with integrated equipment

- Fig. 7 shows the refrigerant circuit in the lg p, h diagram with two-stage evaporator of the "patent"

- Fig. 8 shows the refrigerant circuit with two-stage evaporator of the "patent"

- Fig. 9 shows the refrigerant circuit in the lg p, h diagram with two-stage evaporator of the "patent" with integrated equipment

- Fig. 10 shows the refrigerant circuit in the lg p, h diagram with two-stage evaporator of the "patent" with integrated equipment and two-stage supercooling (and deheater)

- Fig. 11 shows the refrigerant circuit with evaporator and measured value combinations (example)

- Fig. 12 shows the legend for the points from the drawings

Embodiment of the invention:

A refrigeration system substantially comprising one or more:

Liquefiers (2), evaporators (4), IHEs (5), refrigerant compressors (1), injection valves (3), refrigerants, refrigeration auxiliary substances and oil.

A refrigeration system, depending on its application, optionally also has one or more of the abovementioned components and, in addition deheaters (24), one or more waste heat utilization exchangers, further supercoolers (25), viewing windows (7), driers (6), filters, valves (8), safety equipment, shut-off equipment, accumulators, oil pumps, distribution systems, electrical and control parts, refrigeration auxiliary substances, etc.

When fitting the injection valve (3) upstream of the evaporator (4), the measured value for limiting suction vapor is taken off at the suction line leading to the refrigerant compressor (1). The measured values for the refrigerant liquid temperature (11) and the evaporator entry pressure (12) are used to control the evaporation (17, 19).
Alternatively, the measured values for the high pressure (22) upstream of the injection valve (3) and for the suction vapor pressure (12) downstream of the injection valve (3), as well as the hot-gas temperature (15) downstream of the compressor (1) or the oil temperature (16) of the latter, are likewise available for controlling the evaporator (4) with downstream IHE (5).